[redacted]

Copy 6
RM E54F03

NACA RM E54F03

NACA

RESEARCH MEMORANDUM

INVESTIGATION OF EFFECT OF INCREASED DIFFUSION OF ROTOR-BLADE SUCTION-SURFACE VELOCITY ON PERFORMANCE OF TRANSONIC TURBINE

By Robert Y. Wong, Daniel E. Monroe, and William T. Wintucky

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

CLASSIFICATION CHANGED

UNCLASSIFIED

To. ____________

NACA Res abs effective

By authority of YRN-128 Date June 24, 1958

AMT 8-12-58

CLASSIFIED DOCUMENT

This material contains information affecting the National Defense of the United States within the meaning of the espionage laws, Title 18, U.S.C., Secs. 793 and 794, the transmission or revelation of which in any manner to an unauthorized person is prohibited by law.

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WASHINGTON

August 24, 1954

C[redacted]AL

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

INVESTIGATION OF EFFECT OF INCREASED DIFFUSION OF ROTOR-BLADE SUCTION-SURFACE VELOCITY ON PERFORMANCE OF TRANSONIC TURBINE

By Robert Y. Wong, Daniel E. Monroe, and William T. Wintucky

3303

SUMMARY

The results of an experimental investigation of a transonic turbine designed for a diffusion parameter of 0.30 are presented herein. The experimental performance of this turbine was compared with the experimental performance obtained for another transonic turbine designed for a diffusion parameter of 0.15. The higher-diffusion turbine had a 4-point loss in design-point efficiency as compared with that of the lower-diffusion turbine. The loss patterns of both turbine configurations, as indicated by surveys, were found to have similar trends. In the region of the hub and mean sections, the efficiency was found to be of comparable levels for both turbines, while from the mean to the tip, a much greater drop-off in efficiency occurred for the higher-diffusion turbine than for the lower-diffusion turbine. From these surveys it is believed that the three-dimensional characteristics of the turbine may have an important effect on rotor losses by causing a transport of low-momentum fluids from the hub and mean sections to the tip region, which add to the measured losses and may even create further loss in this region.

INTRODUCTION

A research program is in progress at the NACA Lewis laboratory to study problems associated with attaining efficient transonic turbines. A transonic turbine is defined as a turbine designed for a rotor-hub-inlet relative Mach number of approximately unity. If efficiencies comparable with conventional turbines can be obtained for transonic turbines, a more compact turbine component will be available for use in turbojet engines. The investigation of one transonic turbine (ref. 1) indicated that, with proper design considerations, turbines designed for rotor-hub-inlet relative Mach numbers of unity can operate with efficiencies of at least 0.85.

[redacted]

Results of recent compressor loss investigations (ref. 2, e.g.)
indicate that diffusion of blade-surface velocities is a very important
design consideration in minimizing compressor-blade losses and that above
a certain value of diffusion the losses increase rapidly. Thus diffusion
should also be considered in the design of turbines in order that blade
losses be minimized. However, to date, the degree to which this param-
eter affects turbine losses has not been evaluated. For example, the
turbine of reference 1, hereinafter designated as transonic turbine A,
was designed for a rotor suction-surface diffusion parameter of 0.15
(diffusion parameter D is defined as the difference between the peak 3303
suction-surface relative velocity and the blade outlet relative velocity
divided by the peak suction-surface relative velocity), but it is not
known whether this value of the diffusion parameter resulted in low or
excessive blade losses. Thus, in order to gain an insight into the
effect of increased diffusion on the performance of transonic turbines,
the investigation of another transonic turbine, hereinafter referred to
as transonic turbine B, designed for a diffusion parameter D of 0.30,
or twice that of transonic turbine A, was conducted.

The results of this investigation are presented herein. Comparisons between the performance of the two turbines were made in order to evaluate the effects of increased blade-surface diffusion on turbine performance. Local adiabatic efficiencies across the turbine from stator inlet to rotor outlet, obtained from surveys, are also presented to indicate where losses resulting from high diffusion manifest themselves.

SYMBOLS

The following symbols are used in this report:

D	diffusion parameter defined as difference between peak suction-surface relative velocity and blade outlet relative velocity (station 5) divided by peak suction-surface relative velocity
Δh	specific work output, Btu/lb
N	rotative speed, rpm
p	absolute pressure, lb/sq ft
r	radius, ft
U	blade velocity, ft/sec
V	absolute gas velocity, ft/sec

W relative gas velocity, ft/sec

w weight flow, lb/sec

γ ratio of specific heats

δ ratio of turbine-inlet total pressure to NACA standard sea-level pressure p_0'/p^*

ϵ function of γ, $\left(\frac{\gamma^*}{\gamma}\right)\left[\frac{\left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{\gamma-1}}}{\left(\frac{\gamma^*+1}{2}\right)^{\frac{\gamma^*}{\gamma^*-1}}}\right]$

η_l local adiabatic efficiency based on total state measurements upstream of stator and downstream of rotor

η_t adiabatic efficiency defined as ratio of turbine work based on torque, weight flow, and speed measurements to ideal work based on inlet total temperature, and inlet and outlet total pressure, both defined as sum of static pressure plus pressure corresponding to the absolute gas velocity

η_x adiabatic efficiency defined as ratio of turbine work based on torque, weight flow, and speed measurements to ideal work based on inlet total temperature, and inlet and outlet total pressure, both defined as sum of static pressure plus pressure corresponding to axial component of absolute gas velocity

θ_{cr} squared ratio of critical velocity at turbine inlet to critical velocity at NACA standard sea-level temperature $(V_{cr,0}/V_{cr}^*)^2$

Subscripts:

cr conditions at Mach number of unity

t tip

x axial direction

0 station upstream of stator (see fig. 1)

1 station at throat of stator passage

2 station at outlet of stator just upstream from trailing edge

3 station at free-stream condition between stator and rotor

4 station at throat of rotor passage

5 station at outlet of rotor just upstream from trailing edge

6 station downstream from turbine

Superscripts:

* NACA standard conditions 3303

' total state

TURBINE DESIGN

Design Requirements

As discussed in the INTRODUCTION, transonic turbine B was designed for a diffusion parameter D of 0.30 to investigate the effect of increased diffusion on the performance of transonic turbines. In order to minimize effects of extraneous factors on this investigation, the rotor-inlet conditions and the peak surface velocities of turbine A were specified as rotor-blade requirements of turbine B. Thus the effect of the stators on losses of the two rotors may be expected to be comparable, and shock losses within the rotor resulting from the high surface velocities may also be expected to be comparable. These blade and diffusion requirements resulted in a turbine B work output approximately 11 percent less than that of turbine A. The design requirements that meet the above stated conditions for the 14-inch cold-air turbine investigated herein are as follows:

$$\Delta h'/\theta_{cr} = 20.2 \text{ Btu/lb}$$

$$\epsilon w \sqrt{\theta_{cr}}/\delta = 11.95 \text{ lb/sec}$$

$$U_t/\sqrt{\theta_{cr}} = 597 \text{ ft/sec}$$

Design Velocity Diagrams

The design velocity diagrams were constructed for the free-stream stations 0, 3, and 6 at the hub, mean, and tip sections and were based on the following assumptions:

(1) Free vortex flow

(2) Simple radial equilibrium

(3) A 3-percent total-pressure loss across the stator

(4) Adiabatic efficiency η_t of 0.88 to obtain velocity diagrams at station 6

3303

These diagrams, together with a sketch of a typical blade channel showing the station nomenclature used, are shown in figure 1. Also included are the velocity diagrams at stations 2 and 5 (just inside the blade passage at the trailing edge), which are used only in the blade design procedure. The assumptions by which these velocities are obtained are fully discussed in the section TURBINE DESIGN of reference 1. Since the rotor-inlet velocity diagrams are identical with those used in the design of transonic turbine A, the exit diagrams differ because of the specified diffusion and peak surface velocity. The turbine discharge is designed for zero whirl, and considerable negative reaction across the hub exists $((W/W_{cr})_3 = 1.000$ and $(W/W_{cr})_6 = 0.753)$. The design reaction (defined as the ratio of rotor-inlet static pressure to rotor-outlet static pressure) across the rotor hub is 0.75 as compared with 0.91 for transonic turbine A. The design gas turning across the rotor varies from 92.4° at the rotor hub to 70.9° at the tip, whereas the gas turning across the rotor of transonic turbine A was 96.7° to 71.7° from hub to tip, respectively. The rotor of transonic turbine B is not designed to choke since the relative critical velocity ratio at station 5 $(W/W_{cr})_5$ varies from 0.803 to 0.914 from hub to tip, respectively.

Rotor-Blade Design

The design procedure used to obtain the rotor blades is described in reference 1. With the specified limit of 1.33 imposed on the surface Mach number and a prescribed sinusoidal variation of the hub midchannel relative critical velocity ratio from inlet to outlet, 27 blades were required, thus resulting in solidities (based on chord) of 2.65 and 1.92 at the hub and tip, respectively. The solidity of turbine B was therefore 24 percent less than that of turbine A.

The rotor-blade coordinates and sections for the hub, mean, and tip are shown in table I and figure 2, respectively. As was found for transonic turbine A (ref. 1), the hub flow channel of transonic turbine B also diverges from inlet to outlet, the mean-section flow channel is approximately a constant area from inlet to outlet, and the tip-section flow channel converges from inlet to outlet. A photograph of the rotor assembly used in this investigation is shown in figure 3.

The design surface velocity distributions for the hub, mean, and tip sections are given in figure 4. The prescribed sinusoidal hub midchannel velocity variation is also indicated. The midchannel velocity

[redacted]

variation required at the mean and tip sections to satisfy both radial equilibrium and the prescribed hub midchannel velocity distribution is also shown; and these midchannel velocities are seen to accelerate rapidly over the first half of the blade, reach a maximum at about midchord, and then remain approximately constant to the outlet. The diffusion parameter D calculated from the critical velocity ratios given in figure 4 was found to be approximately a constant value of 0.30 from hub to tip.

APPARATUS, INSTRUMENTATION, AND METHODS

The apparatus, the instrumentation, and the methods used in calculating the performance parameters used in this investigation were the same as those described in detail in reference 1. A schematic drawing of the apparatus is shown in figure 5.

The experimental investigation was conducted by operating the turbine at constant nominal inlet conditions of 32 inches mercury absolute and 145° F and at constant speed values over a range of 30 to 130 percent of design speed in even increments of 10 percent. For each speed investigated, a range of total-pressure ratio from approximately 1.4 to that corresponding to limiting loading was obtained. Detailed circumferential and radial surveys of total temperature and total pressure were made downstream of the rotor (fig. 5, station 6) at approximately design speed and work output.

The precision of the measurements used in calculating the over-all performance parameters is estimated to be within the following limits:

Temperature, °F	±0.5
Pressure, in. Hg	±0.05
Turbine speed, rpm	±10.0
Torque, percent of design	±0.5

The maximum probable error in the adiabatic efficiency at or near design-point operation was estimated to be within ±0.5 point. The reproducibility of a given set of data on the automatic curve tracer used only to record survey data was observed to be within ±0.5 percent.

RESULTS

The results of the experimental investigation of transonic turbine B, designed for a diffusion parameter D of 0.30, are presented herein and compared with the performance of transonic turbine A, designed for a diffusion parameter of 0.15, to indicate the effects of increased diffusion on the performance of transonic turbines. Radial and circumferential surveys downstream of the rotor are also presented to further indicate the effects of the increased diffusion on turbine performance.

Over-All Performance

The over-all performance of transonic turbine B is presented in figure 6. The equivalent specific work output $\Delta h'/\theta_{cr}$ is shown as a function of the weight flow - speed parameter $\varepsilon wN/\delta$, with rating total-pressure ratio $p_0'/p_{6,x}'$, percent design speed, and adiabatic efficiency η_x as contours. Design equivalent specific work output was obtained at design speed with an adiabatic efficiency η_x of approximately 0.81. Zero exit whirl was observed to occur at approximately design-point operation, as designed; thus the adiabatic efficiency η_x is equivalent to the adiabatic efficiency η_t at the design-point operation. As reported in reference 1, design work output at design speed was obtained from transonic turbine A at an adiabatic efficiency η_t of 0.85. Thus, there was a 4-point loss in the design-point efficiency of turbine B as compared with turbine A.

Survey Investigation

In order to gain a further insight into the loss characteristics associated with increased diffusion of the suction-surface velocity, detailed radial and circumferential surveys were made downstream of the rotor at approximately design-point operation. The results of the survey investigation are shown in figure 7 in terms of local adiabatic efficiency across the turbine from the stator inlet to the turbine outlet. As pointed out in reference 1, these results serve only to indicate general trends in local adiabatic efficiency and may not be representative of the actual level. An inspection of figure 7 indicates that there is a region of low local efficiency near the hub; also there appears to be a region of low efficiency extending from hub to tip which divides the regions of high efficiency. These regions of low local adiabatic efficiency have been observed to be effects of the stator passage vortex and the stator-blade wakes, respectively (ref. 3). Thus the stator effects are superimposed on the rotor effects. In order to separate the rotor effects from the stator effects, a plot of peak local adiabatic efficiency against radius is presented in figure 8. A similar plot for transonic turbine A is also shown; these results were obtained with instrumentation having characteristics that are considered more comparable with those used in the survey investigation of transonic turbine B. Thus the peak values of local adiabatic efficiency given herein for transonic turbine A are slightly different from those given in figure 10(b) of reference 1. It should be noted, however, that the general trends reported in reference 1 are valid. An inspection of figure 8 shows that the local adiabatic efficiencies in the region of the hub and mean of the two turbine configurations are of comparable levels. For both turbines the efficiency drops off rapidly from about the mean radius to the tip, with a much steeper gradient in efficiency in the region of the tip for transonic turbine B. Thus it is indicated that the increased losses associated with the increased diffusion parameter are manifesting themselves at the turbine exit in the tip region only.

DISCUSSION

The results of the over-all performance investigation of turbine B indicate a 4-point loss in efficiency as compared with turbine A. In 3303
evaluating this loss, consideration must be given to the 11-percent difference in work output between the two turbines (see TURBINE DESIGN section). For comparable total losses for both turbines, this reduction in work output would result in approximately $1\frac{1}{2}$ points reduction in efficiency. Thus, even taking this effect into account, a considerable increase in the total losses of turbine B is incurred as compared with turbine A. Although it appears that the increase in total losses is caused by the direct effect of increased diffusion on blade viscous losses, it must be remembered that the three-dimensional characteristics of the turbine can also combine with this diffusion effect to complicate the mechanism through which the losses occur. The results of the survey investigation described herein and presented in figure 8 can be used to illustrate some of these three-dimensional effects.

The measured over-all efficiencies at the hub and mean sections were of comparable levels for both turbines A and B. Because of three-dimensional effects, the over-all losses measured at any particular radial station may be dependent upon the transport of any low-momentum fluids to or from this station. It is believed that centrifugal force acting on the boundary-layer fluids of the hub and mean sections causes a movement of a certain amount of these fluids into the vicinity of the tip. From the surveys presented herein, it appears that, if there was any increase in over-all losses at the hub and mean sections of transonic turbine B, these increased losses were being measured in the region of the tip as a result of bleedoff of low-momentum fluids by centrifugal force. This bleedoff could have resulted in the comparable measured efficiencies at the hub and mean sections at the turbine exit. It is also felt that this bleedoff or asperating effect would tend to retard possible flow separations resulting from high diffusion at these two sections, thereby tending to improve the loss characteristics of these sections.

At the tip, the surveys indicated that the efficiency of transonic turbine B dropped off considerably from that of transonic turbine A. Since the tip section of transonic turbine B was also designed for a high diffusion as compared with that of transonic turbine A, large increases in losses would be expected because of the effects of the low-momentum fluids moving into the tip region. These effects will not only add to the measured losses in this region; but, if appreciable movement takes place on the blade surfaces, these low-momentum fluids may also combine with local tip losses occurring as a result of complex tip secondary flows and high diffusion to further disrupt the flow in the critical tip region, causing even greater losses than that due to the increased diffusion alone.

From the foregoing considerations, it is evident that, although the loss in efficiency of transonic turbine B from that of transonic turbine A is attributable to the effects of the increased design diffusion, the three-dimensional characteristics of the turbine can not only have a considerable effect on the mechanism through which these losses occur but also make difficult a rigorous evaluation of the effect of increased diffusion on losses.

SUMMARY OF RESULTS

The results of the experimental investigation of a transonic turbine designed for a diffusion parameter of 0.30 are compared with the experimental performance of another transonic turbine designed for a diffusion parameter of 0.15 in order to evaluate the effect of increased diffusion on the performance of transonic turbines. The pertinent results of this investigation can be summarized as follows:

1. At design speed, design equivalent specific work was obtained at an adiabatic efficiency of 0.81. This represented a decrease of 4 points in efficiency from the transonic turbine designed for a lower-diffusion parameter.

2. The loss patterns of both turbine configurations, as indicated by surveys taken downstream of the turbine rotor, were found to have similar trends. In the region of the hub and mean sections, the measured losses were found to be of comparable levels, while in the region from the mean to the tip, a much greater drop-off in measured efficiency occurred for the higher-diffusion turbine than for the lower-diffusion turbine. From these surveys it is believed that the three-dimensional characteristics of the turbine may have an important effect on rotor losses by causing a transport of low-momentum fluids from the hub and mean sections to the tip region, which add to the measured loss in this region and may even create further losses in this region.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, June 4, 1954

REFERENCES

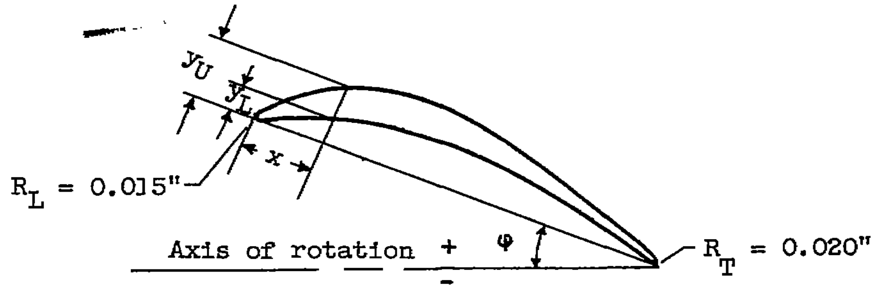
1. Stewart, Warner L., Wong, Robert Y., and Evans, David G.: Design and Experimental Investigation of Transonic Turbine with Slight Negative Reaction Across Rotor Hub. NACA RM E53L29a, 1954.

2. Lieblein, Seymour, Schwenk, Francis C., and Broderick, Robert L.: Diffusion Factor for Estimating Losses and Limiting Blade Loadings in Axial-Flow-Compressor Blade Elements. NACA RM E53D01, 1953.

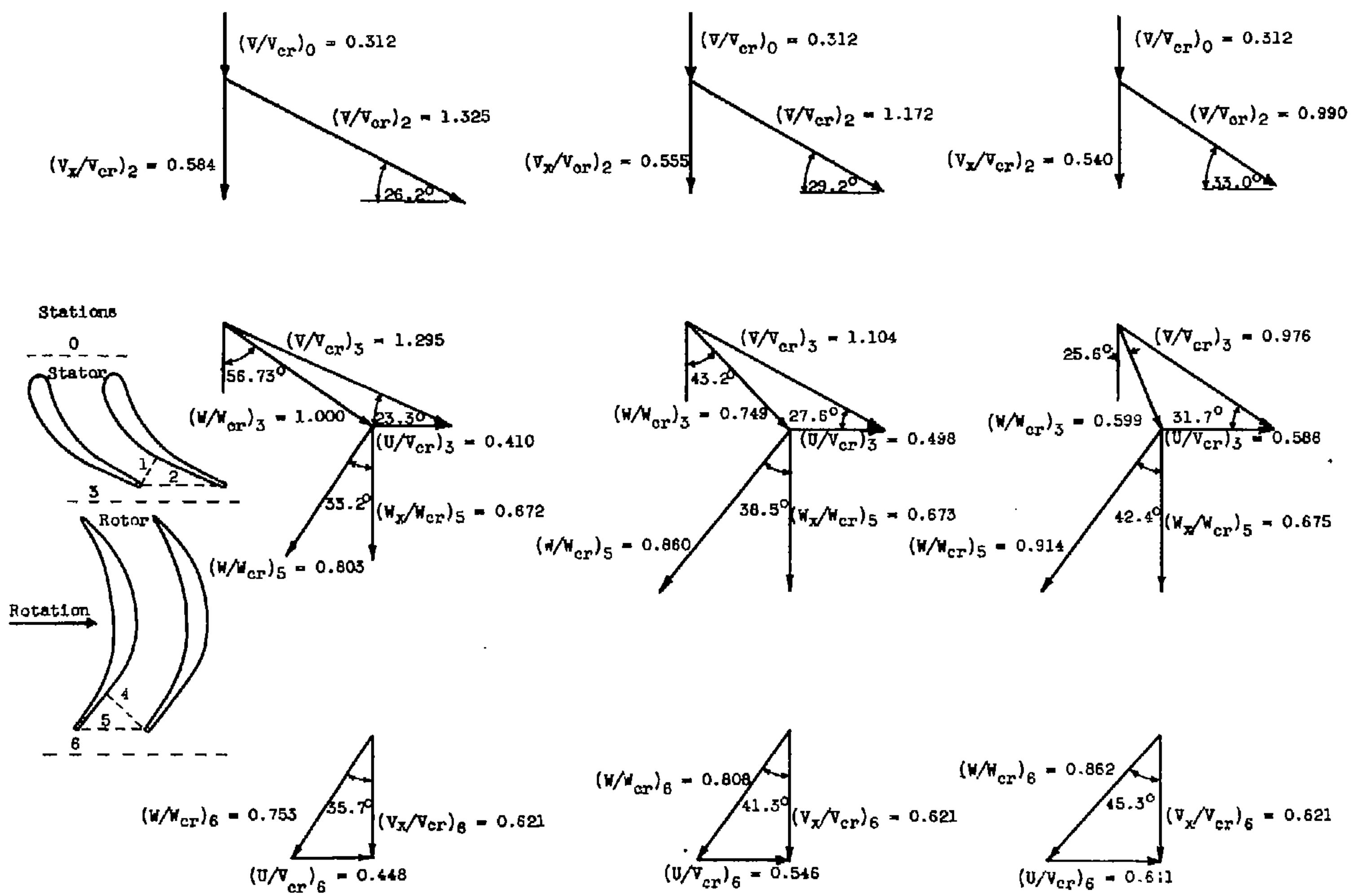
3. Whitney, Warren J., Buckner, Howard A., Jr., and Monroe, Daniel E.: Effect of Nozzle Secondary Flows on Turbine Performance as Indicated by Exit Surveys of a Rotor. NACA RM E54B03, 1954.

3303

TABLE I. - ROTOR-BLADE-SECTION COORDINATES

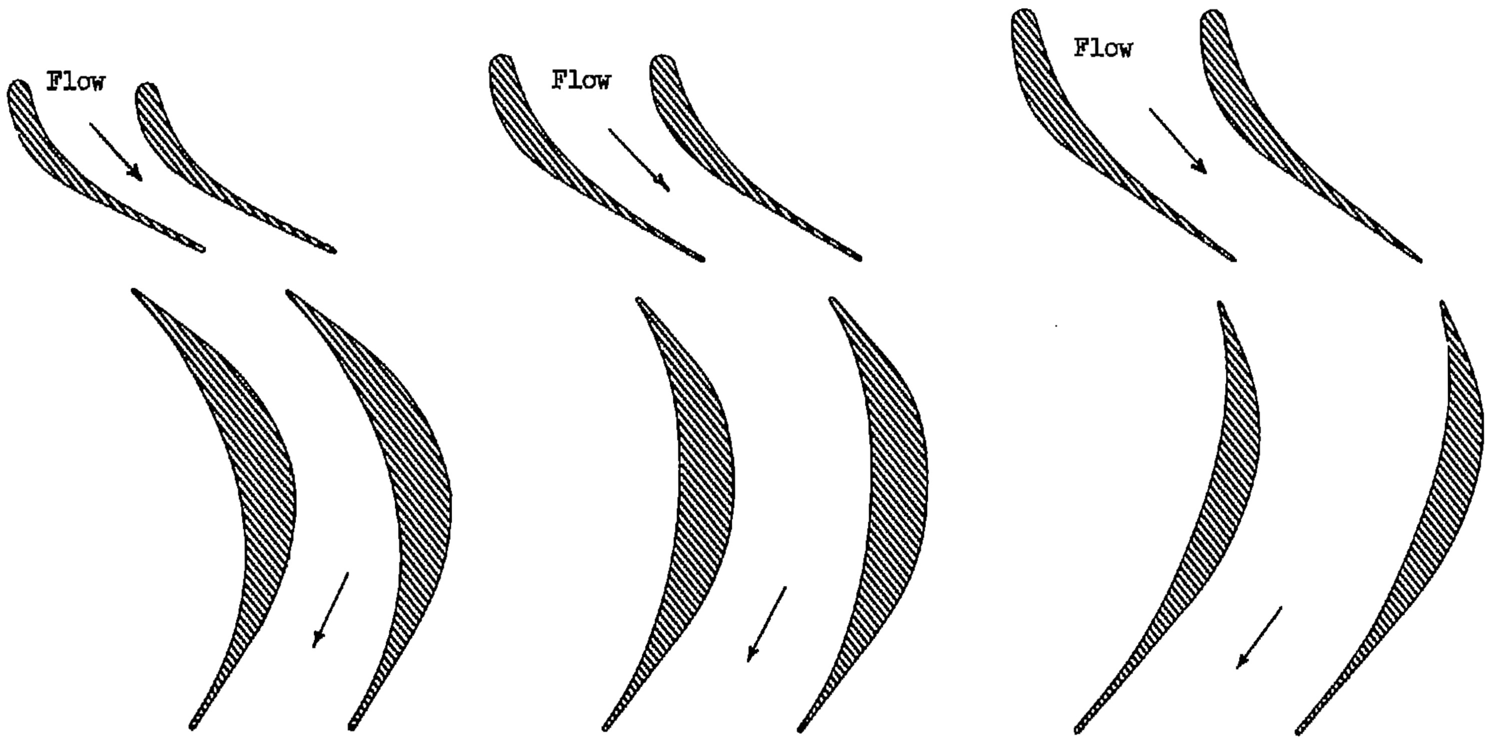


	Hub		Mean		Tip	
φ, deg	-9.8		4.6		20.0	
r/r_t	0.70		0.85		1.00	
x, in.	y_L, in.	y_U, in.	y_L, in.	y_U, in.	y_L, in.	y_U, in.
0	0.015	0.015	0.015	0.015	0.015	0.015
.100	.063	.128	.058	.125	.054	.120
.200	.134	.235	.124	.237	.114	.221
.300	.198	.341	.180	.345	.165	.322
.400	.255	.445	.228	.459	.208	.408
.500	.306	.541	.268	.556	.244	.476
.600	.352	.628	.302	.623	.274	.530
.700	.394	.704	.330	.690	.299	.573
.800	.432	.772	.354	.734	.319	.607
.900	.466	.831	.373	.768	.335	.632
1.000	.496	.880	.389	.792	.348	.650
1.100	.522	.920	.402	.808	.358	.661
1.200	.544	.951	.412	.818	.366	.666
1.300	.562	.972	.419	.823	.371	.665
1.400	.575	.984	.423	.822	.373	.658
1.500	.583	.987	.424	.816	.372	.645
1.600	.587	.981	.422	.804	.368	.627
1.700	.586	.967	.416	.785	.361	.603
1.800	.580	.944	.407	.759	.352	.574
1.900	.568	.913	.395	.725	.340	.539
2.000	.551	.873	.379	.682	.326	.500
2.100	.528	.823	.359	.629	.309	.458
2.200	.499	.763	.335	.562	.290	.415
2.300	.464	.692	.307	.495	.269	.378
2.400	.422	.610	.275	.426	.246	.335
2.500	.372	.520	.239	.360	.220	.294
2.600	.314	.428	.198	.292	.191	.250
2.700	.248	.335	.151	.226	.158	.210
2.800	.173	.243	.098	.158	.122	.168
2.900	.089	.150	.038	.091	.084	.130
2.988	----	----	.020	.020	----	----
3.000	.002	.052	----	----	.044	.088
3.026	.020	.020	----	----	----	----
3.100	----	----	----	----	.003	.047
3.134	----	----	----	----	.020	.020



(a) Hub; r/r_t, 0.70. (b) Mean; r/r_t, 0.85. (c) Tip; r/r_t, 1.00.

Figure 1. - Velocity diagrams at hub, mean, and tip sections for various axial stations.



(a) Hub.   (b) Mean.   (c) Tip.

Figure 2. - Stator- and rotor-blade passages and profiles.

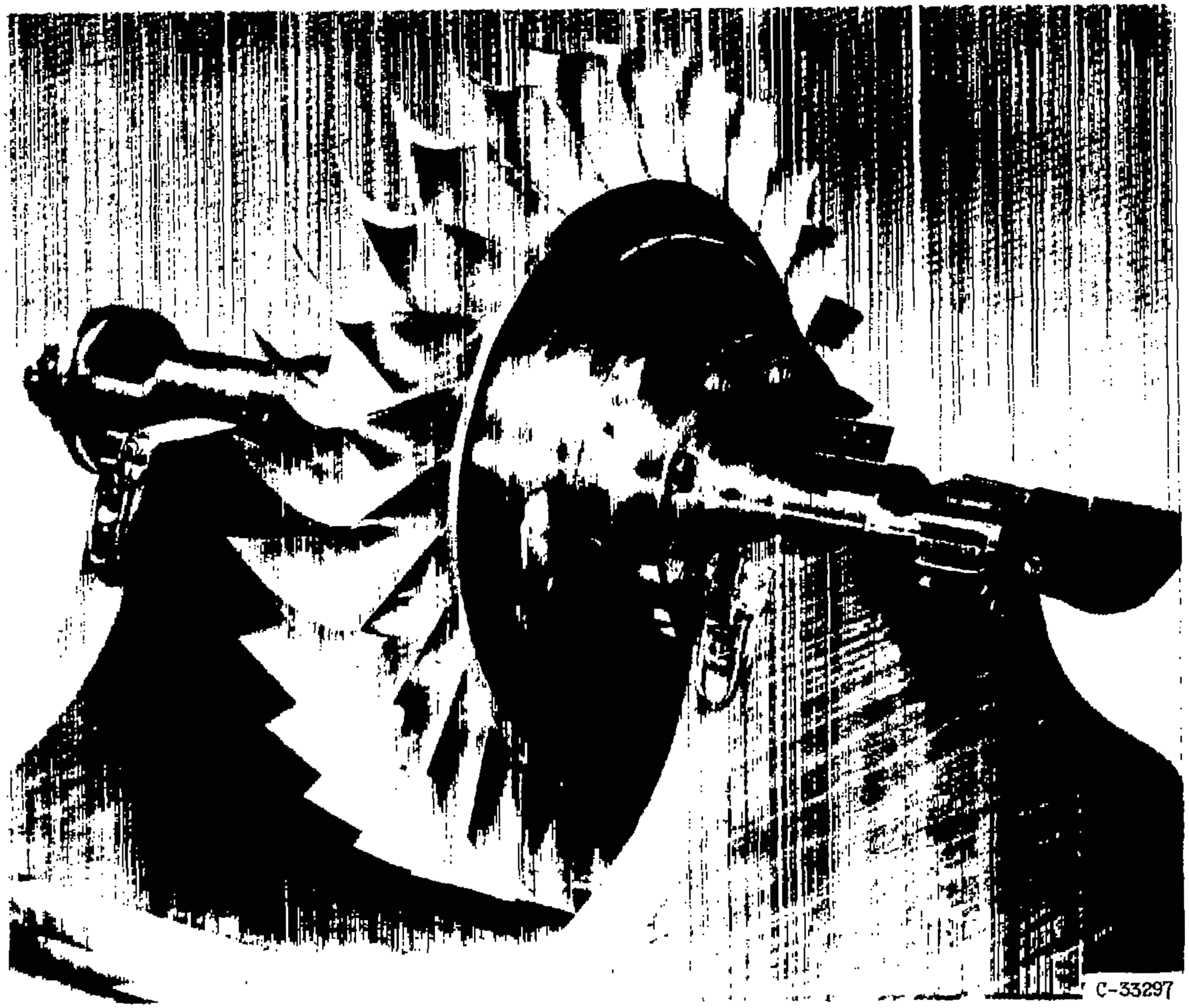


Figure 3. - Photograph of rotor of transonic turbine B.

3303

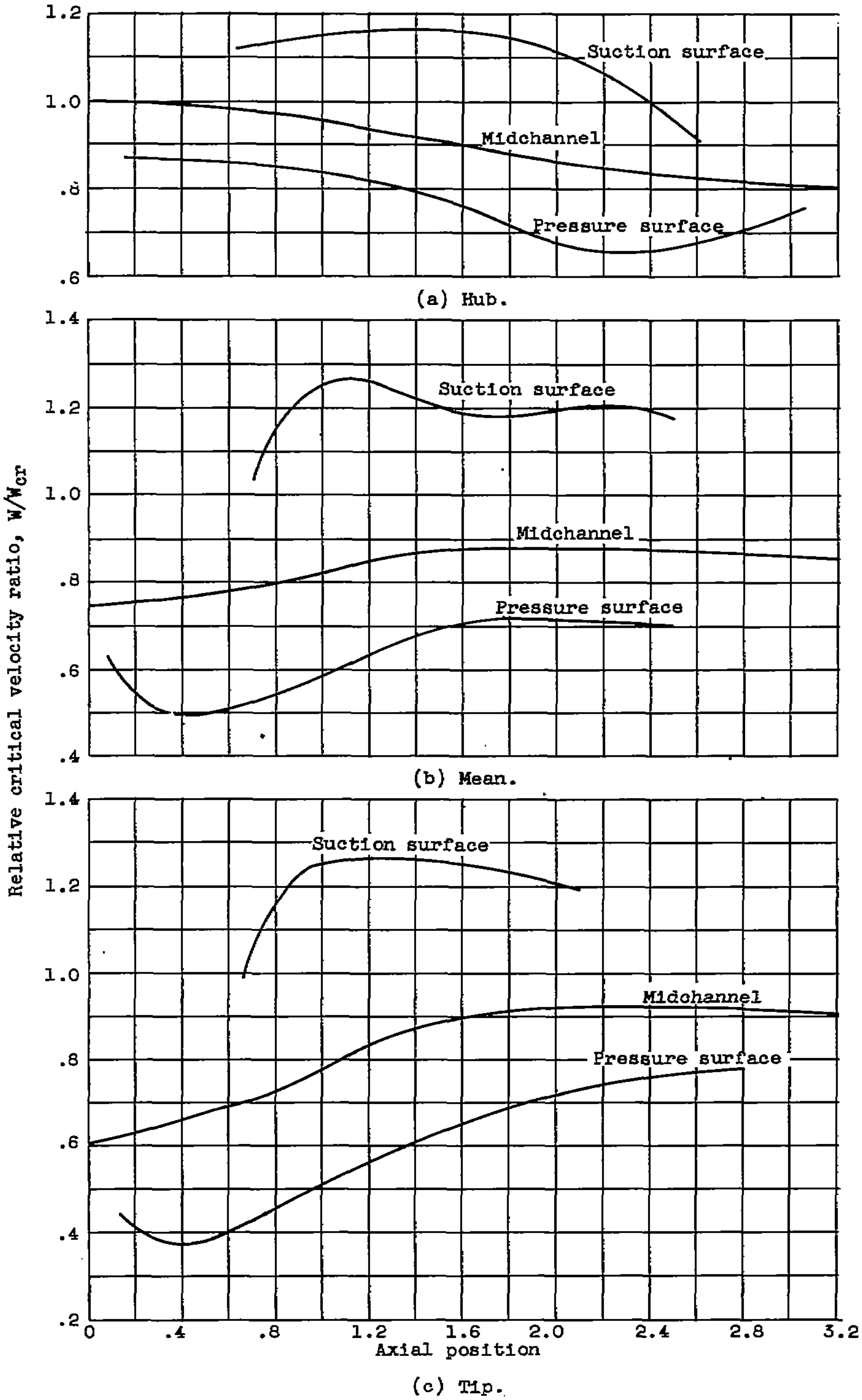


Figure 4. - Design rotor midchannel and surface velocity distributions at hub, mean, and tip.

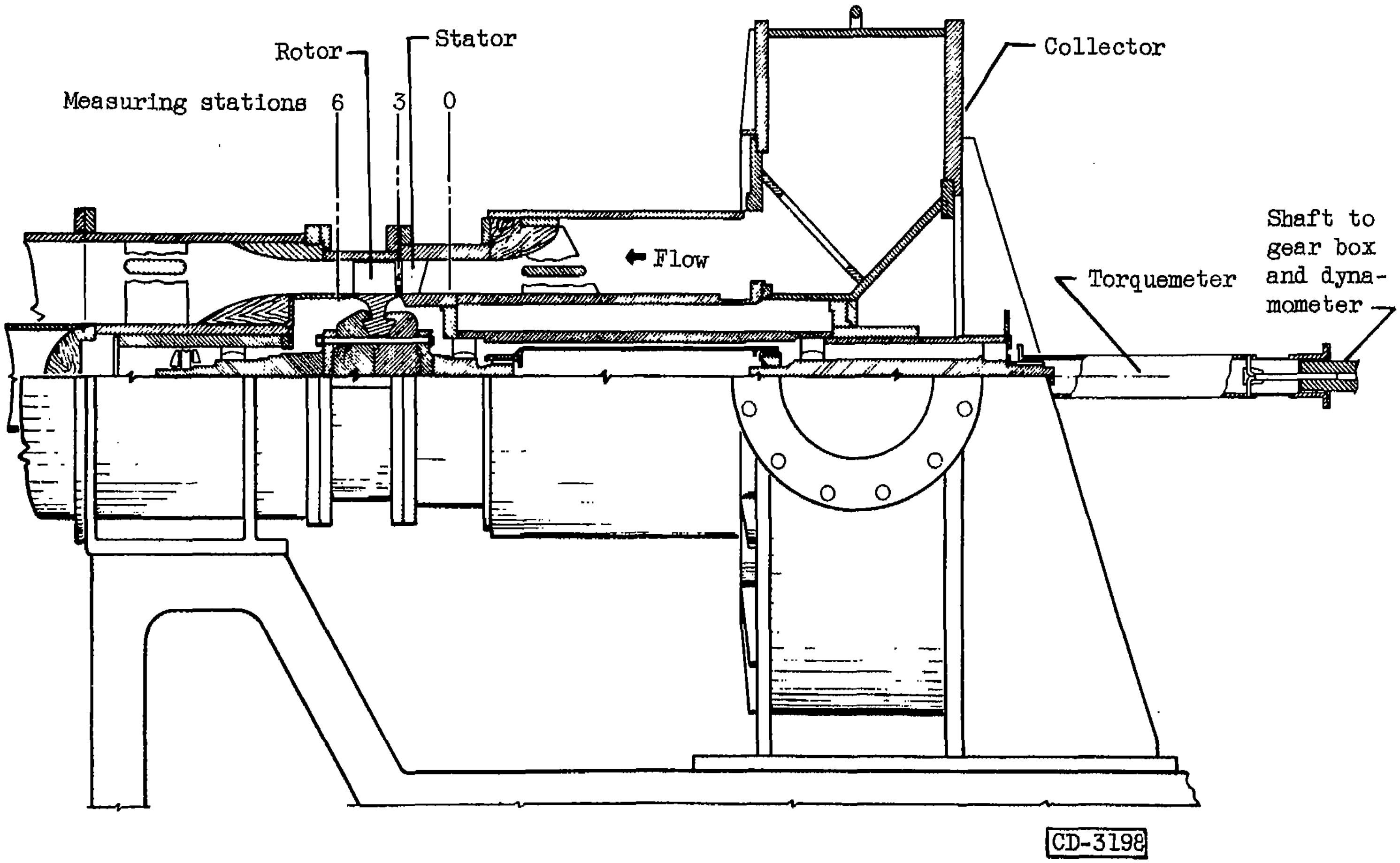


Figure 5. - Diagrammatic sketch of cold-air turbine test section.

3303

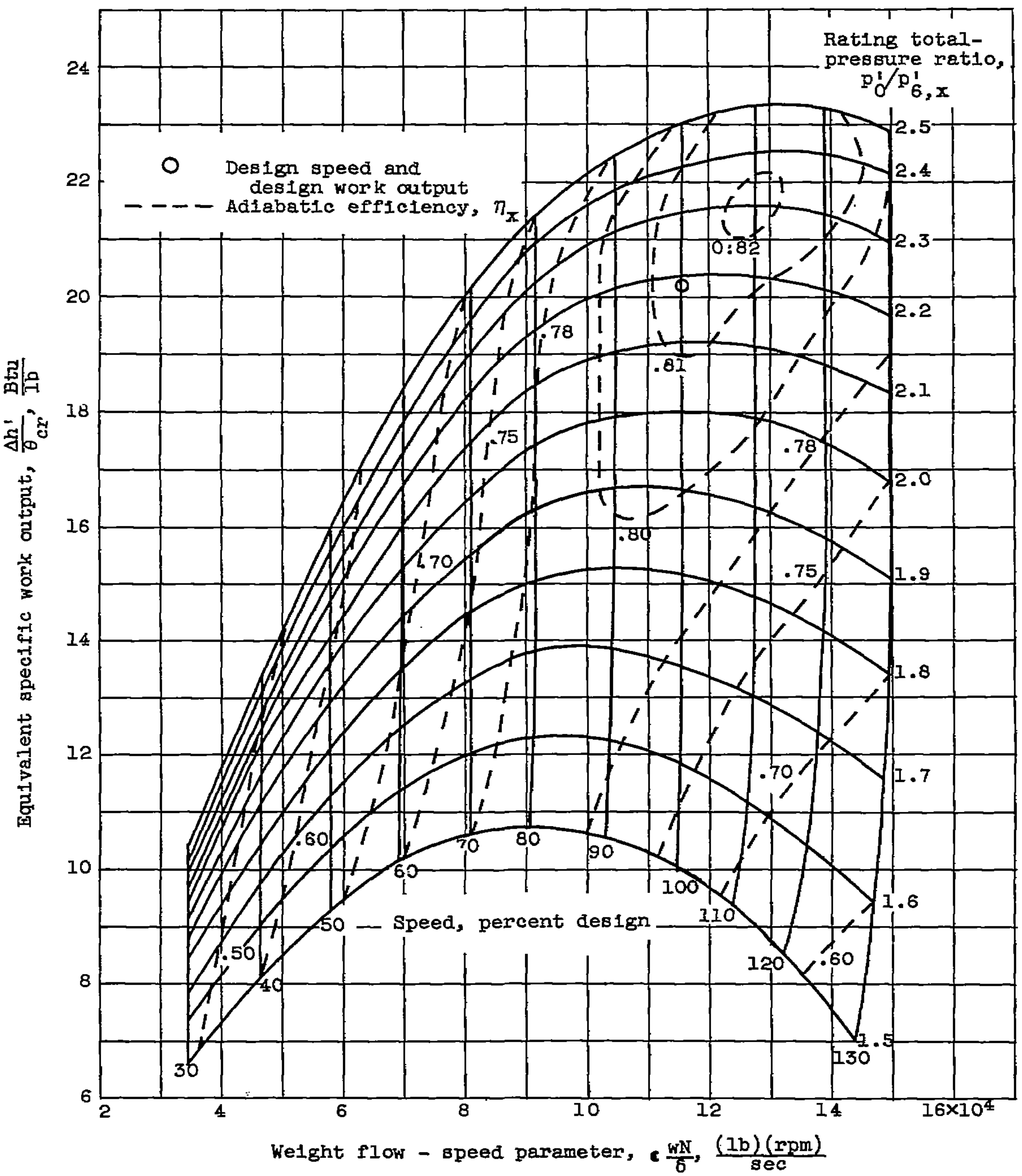


Figure 6. - Experimentally obtained performance map for transonic turbine B.

[redacted]

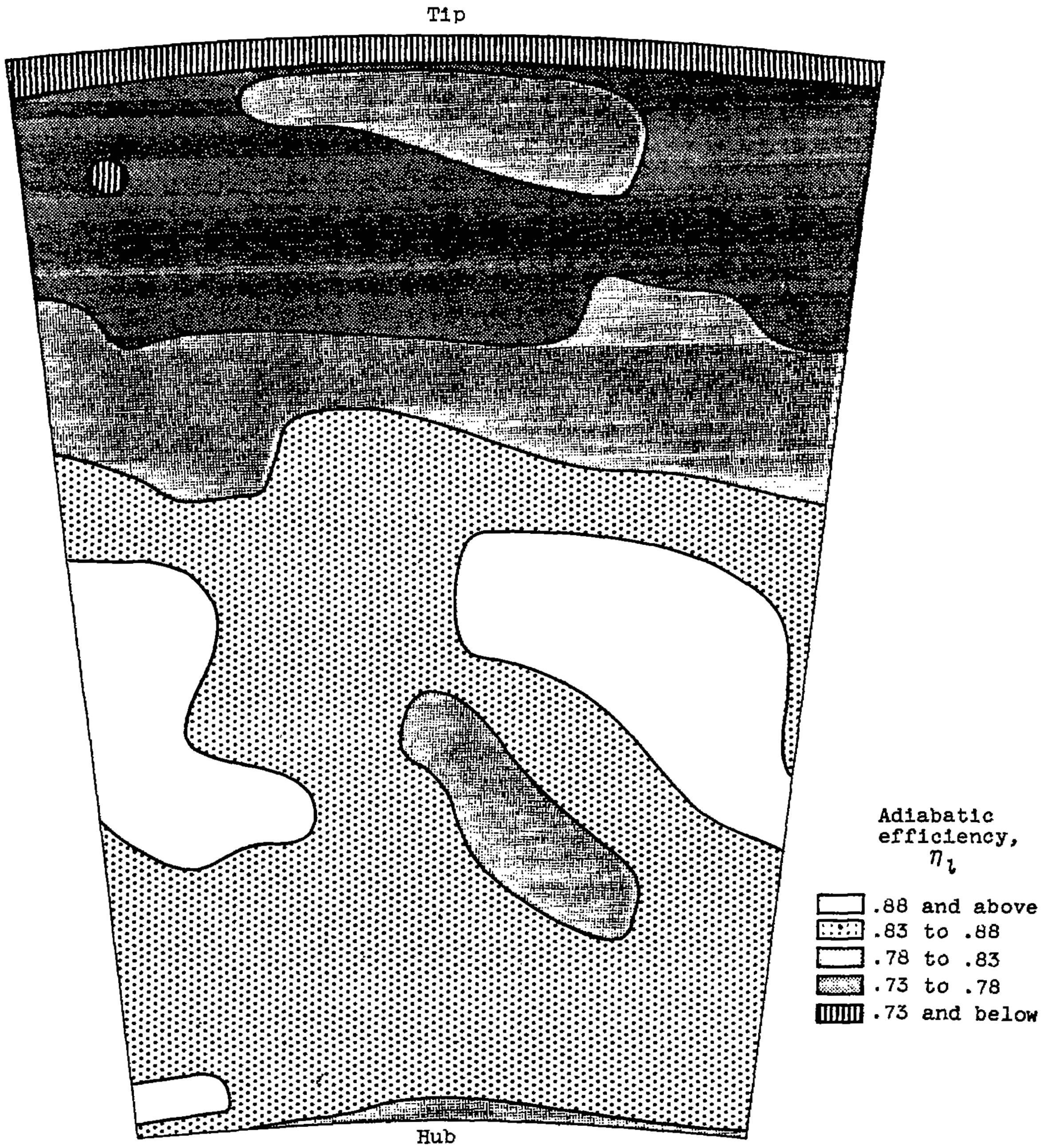


Figure 7. - Survey results behind rotor at design point.

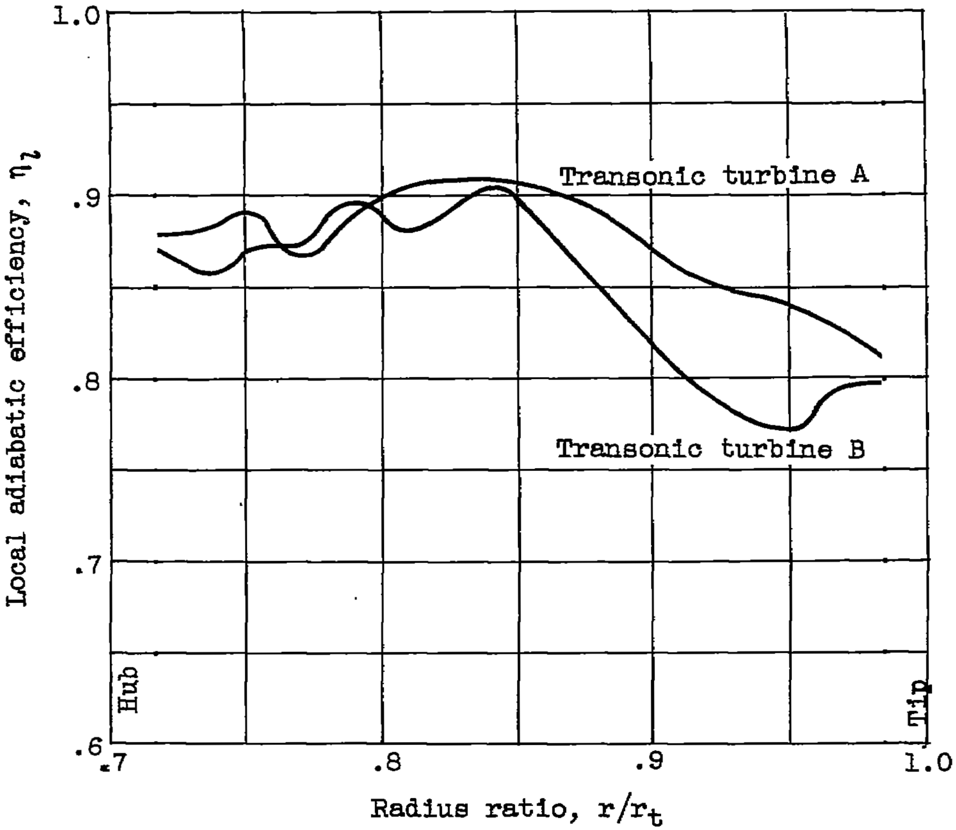


Figure 8. - Radial variation of peak local adiabatic efficiency.

[redacted]